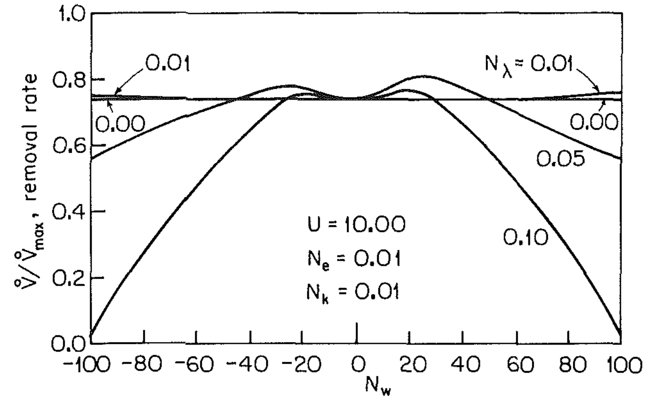

Fig. 5 Material removal rate as a function of beam focusing number

nondimensionalized by the theoretical maximum, i.e., the material removal in the absence of conduction losses:

$$\dot{V}/\dot{V}_{\max} = \frac{2}{\pi}\,(N_e + UN_k)\int_0^\infty S_\infty(\eta)d\eta \tag{6}$$

As expected, the removal rate drops off rapidly for laser beams with their focal points far above or below the surface. In the vicinity of $N_w = 0$ the removal rate is relatively flat with a slight minimum for small negative N_w. Thus it is seen that achieving a deeper groove has its price: A deeper groove has steeper walls with larger total surface area, resulting in larger conduction losses. The observation of a maximum removal rate for a beam focused above the surface has been made experimentally by Wallace [9], who investigated laser shaping of ceramics.

It is interesting to note that it is possible to achieve higher material removal rates for a diverging beam ($N_\lambda > 0$) than for a parallel beam for a few focal point positions above and below the surface. This may be explained by looking at the beam intensity hitting the groove surface, equation (4): For a steep surface the vertical component of the flux onto the wall becomes quite small, so that the contribution of the small radial intensity component (nonexistent for parallel beams) becomes very important.

Conclusions

A heat transfer model for evaporative cutting of a semi-infinite body with a moving continuous wave laser has been developed and solved numerically to investigate the effects of beam focusing and expansion on the size and shape of a groove. It was seen that the depth of the grooves increases and passes through a maximum when the beam is focused slightly inside the material. The groove depth decreases when the beam is focused above the surface of the material. Thus, the groove depths can be increased by using lenses with long focal lengths. Longer focal length lens give larger minimum beam radii at the focal plane, but with a lower beam divergence rate. On the other hand, maximum removal rates are obtained for beams with appreciable divergence rates focused slightly above the surface.

Acknowledgments

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Boiling Incipience in Plane Rotating Water Films

I. Mudawwar[1] and M. A. El-Masri[2]

Nomenclature

a = centrifugal acceleration = $\omega^2 R$
g = gravitational acceleration
h = single-phase convective heat transfer coefficient = $q/(T_w - T_{sat})$
h^* = dimensionless heat transfer coefficient = $(h\,\nu_l^{2/3})/(k_l a^{1/3})$
h_{fg} = latent heat of vaporization
k = thermal conductivity
k_s = wall roughness
Pr = Prandtl number
q = wall heat flux
q_i = incipient heat flux
$\bar{q}_i$ = incipient heat flux defined in equation (6)
R = radius of rotation
Re = film Reynolds number = $4\Gamma/\mu_l$
T = temperature
y_{lam} = thickness of the laminar sublayer
β = dimensionless film thickness = $\delta\, a^{1/3}/\nu_l^{2/3}$
Γ = mass flow rate per unit film width
ΔT_{sat} = wall superheat = $T_w - T_{sat}$
δ = film thickness
μ = dynamic viscosity
ν = kinematic viscosity
ρ = density
σ = surface tension
τ_w = wall shear stress
ω = rotational speed
$\bar{\omega}$ = rotation number = $\omega R^2/\nu_l$

Subscripts

l = liquid
sat = saturated
v = vapor
w = wall

[1]School of Mechanical Engineering, Purdue University, W. Lafayette, IN 47907.

[2]Department of Mechanical Engineering, Massachusetts Institute of Technology, Cambridge, MA 02139.

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Introduction

Knowledge of heat transfer in rotating liquid films is of paramount importance for evaluating the thermal efficiency of gas turbines with water-cooled blades. Centrifugal forces constitute the primary driving forces for liquids flowing in radial rotating channels. Coriolis forces, on the other hand, tend to thin out the flow in the form of a film that covers one side of the channel. Fully developed motion of the film is determined by a balance between centrifugal and shear forces. Thus, rotating film motion resembles that of a free-falling gravity-driven film since both are characterized by a balance between shear and body forces. However, Coriolis forces can strongly influence interfacial waves and turbulent velocity fluctuations of rotating films. This is evident from the results of Kirkpatrick (1980), who compared film thickness measurements for the cases of free-falling and rotating films. His data indicate profound waviness at higher Reynolds numbers for the case of gravity-driven films. On the other hand, interfacial waves in rotating films were found to stabilize at Reynolds numbers in excess of 8000.

This paper focuses on the effects of Coriolis forces and wall roughness on the convective heat transfer coefficient and the incipient boiling heat flux in thin rotating films.

Experimental System

The experimental apparatus used in the present work has been described in detail in the previous paper by Mudawwar et al. (1985). As shown in Fig. 1, the primary rotating system consisted of a 34.30 cm o.d. aluminum disk on which the test channel was mounted. The disk was flanged to the end of a stainless shaft and rotated at speeds up to 1775 rpm by a 7.5 hp motor equipped with a continuous full-range speed controller. A stationary vessel surrounding the rotating disk (not shown in the figure) provided accurate pressure control over the range 1.0–5.41 atm. The deionized water was preheated by external electric heaters before entering the shaft. Another stagnation preheater was installed inside the rotating disk to overcome heat losses from the water to the shaft. The flow was forced through a nozzle into the surface of the preheater. During operation, stagnation preheater power was increased until the saturation temperature was reached. The saturated water was collected inside a spiral pocket and diverted by centrifugal forces into the radial test channel. Radial film flow was established by a nozzle-shaped injection plate. The film was thinned by centrifugal forces to approximately 0.04 mm at the surface of the test heater, the center of which was located 13.0 cm from the shaft axis. The heating module consisted of a nichrome ribbon sandwiched between boron nitride plates and covered with a copper plate, which transferred the heat to the liquid film. Electric power was supplied across the terminals of the ribbon and dissipated to the film over a 12×6.35 mm^2 heat transfer area.

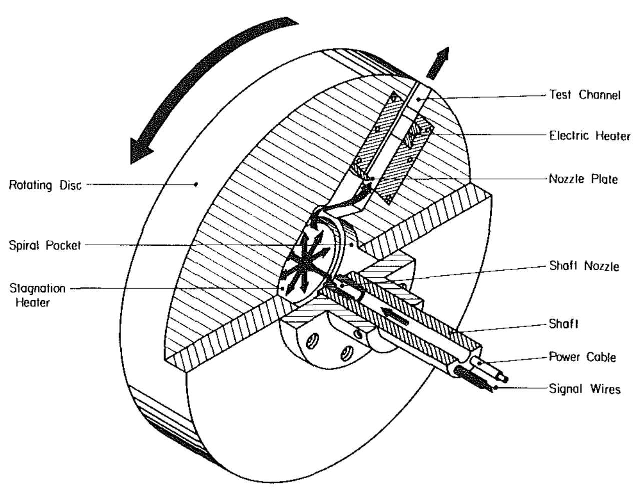

Fig. 1 Sectional diagram of the rotating test section

Results and Discussion

Heat transfer data for rotating films were obtained in the form of boiling curves for a wide range of rotational speeds (ω = 500–1775 rpm), pressures (p = 1.0–5.41 atm), and film Reynolds numbers (Re = 7.1×10^3– 60.7×10^3). Figure 2 shows heat transfer results for several flow rates and a/g = 146. At low pressures the critical heat flux occurred after a relatively small range of boiling heat flux. For a fixed mass flow rate and centrifugal acceleration, higher pressures considerably increased the critical heat flux. Higher acceleration levels increased both the critical heat flux and the heat transfer coefficient prior to boiling.

Following earlier studies (Chun and Seban, 1971) on gravity-driven films, the film thickness δ and the single-phase convective heat transfer coefficient h can be correlated in the following manner:

$$\beta = \frac{\delta\ a^{1/3}}{\nu_l^{2/3}} = f(\mathrm{Re}) \tag{1}$$

$$h^* = \frac{h\ \nu_l^{2/3}}{k_l\ a^{1/3}} = f(\mathrm{Re},\ \mathrm{Pr}_l) \tag{2}$$

where

$$\mathrm{Re} \equiv \frac{4\Gamma}{\mu_l} \tag{3}$$

However, disagreements between free-falling and rotating film thickness correlations have been reported by Kirkpatrick (1980), who failed to develop an empirical equation for his rotating film data according to equation (1). Furthermore, single-phase heat transfer data obtained in the present study were much greater than predicted by Chun and Seban's correlation for free-falling films (i.e., $a = g$) undergoing interfacial evaporation. A possible explanation for the significant difference between gravity-driven and rotating film correlations is the strong influence of Coriolis forces (and possibly wall roughness) on single-phase convection in the film.

Thus, equations (1) and (2) can be modified for rotating films by accounting for Coriolis force effects. That is,

$$\beta = f(\mathrm{Re},\ \hat{\omega}) \tag{4}$$

$$h^* = f(\mathrm{Re},\ \mathrm{Pr}_l,\ \hat{\omega}) \tag{5}$$

The correlation given in Fig. 3 has the same Prandtl number exponent as Chun and Seban's correlation for free-falling films, and possesses an accuracy of $\pm$ 30 percent.

From the analysis of Hsu and Graham (1961), Han and Griffith (1965), and Bergles and Rohsenow (1964), nucleation is believed to commence on a heating surface when the liquid surrounding a growing bubble exceeds the temperature of its outer surface. In the presence of a wide range of cavity sizes, nucleation of a hemispherical bubble within a superheated wall layer characterized by a linear temperature distribution is determined by the following equation:

$$\bar{q}_i = \frac{8 T_{\mathrm{sat}} \sigma}{k_l \rho_v h_{fg}}\ h^2 \tag{6}$$

The experimental results for the incipient boiling heat flux in rotating films are compared in Fig. 4 to equation (6). The ratio $q_i/\bar{q}_i$ is far from unity for most of the operating conditions. Such inconsistencies have been attributed by many researchers to the condition of the boiling surface. If the boiling surface is very smooth, for example, equation (6) can be in great error since nucleation can be totally suppressed. Davis

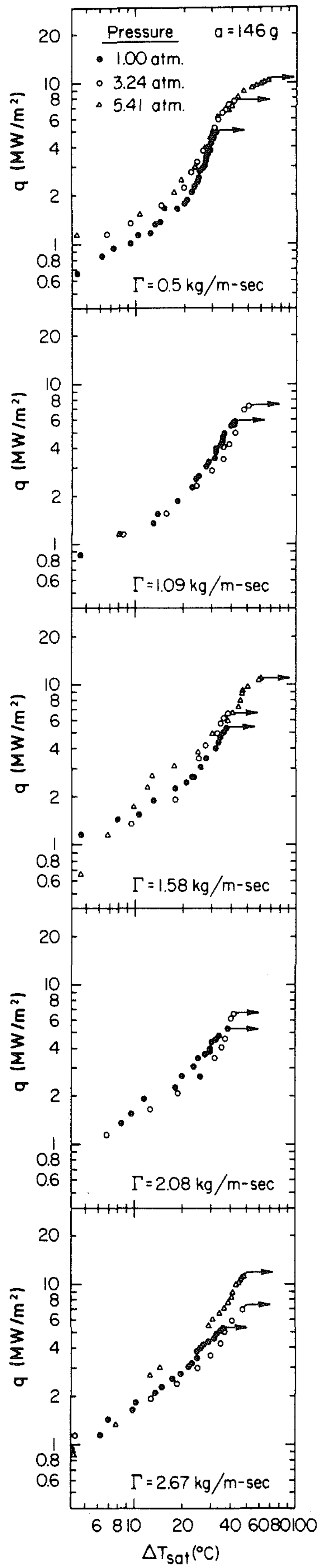

Fig. 2 Boiling heat transfer data for $a/g = 146$

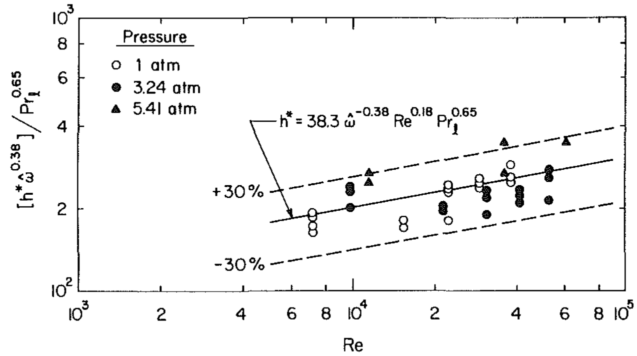

Fig. 3 Variation of the dimensionless single-phase evaporation heat transfer coefficient with Reynolds number and rotation number

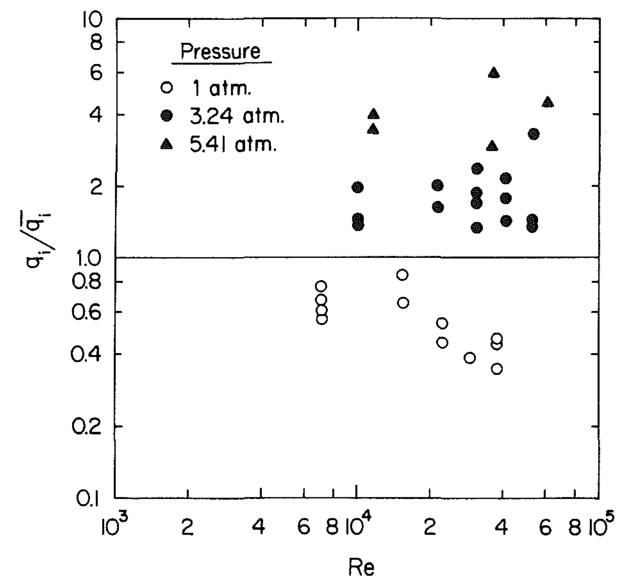

Fig. 4 Comparison between experimental incipient heat flux data and equation (6)

and Anderson (1966) modified equation (6) to account for the characteristics of the boiling surface. Nevertheless, their model was also based on the assumption of linear temperature distribution in the vicinity of the wall. Thus the validity of equation (6) is dependent on the existence of a laminar sublayer within the thermal boundary layer. In the range of operating conditions of the rotating water film data ($Pr \approx 1$) of the present study, the thickness (y_{lam}) of the laminar sublayer can be approximated by

$$\frac{y_{lam}\sqrt{\dfrac{\tau_\omega}{\rho_l}}}{\nu_l}=5 \tag{7}$$

For a fully developed rotating film, equation (7) reduces to

$$\frac{y_{lam}}{\delta}=\frac{5}{\beta^{1.5}} \tag{8}$$

The surface can be considered hydraulically smooth or rough depending on the ratio of the wall roughness k_s to the thickness of the laminar sublayer. That is,

$$\text{smooth surface: } \frac{k_s}{\delta}<\frac{5}{\beta^{1.5}} \tag{9}$$

$$\text{rough surface: } \frac{k_s}{\delta}>\frac{70}{\beta^{1.5}} \tag{10}$$

Thus equation (6) can only be valid if the condition of equation (9) is satisfied. By comparison with our rotating film data, the smooth wall condition corresponds to $k_s < 1\ \mu$m (based on the rotating film thickness data of Kirkpatrick, 1980). Since the heat transfer surface was fabricated by milling, most of the data were found to fall in the roughness range corresponding to equation (10).

Conclusions

This study has focused on boiling heat transfer to thin rotating water films. The primary objective was to study the effects of pressure, centrifugal acceleration, and flow rate on boiling incipience. The primary conclusions are as follows:

1 Acceleration increased the convective heat transfer coefficient prior to boiling and delayed boiling incipience.

2 Coriolis forces play a significant role in rotating film convective processes. Prior to boiling, these forces influence the turbulent velocity fluctuations within the film as well as the stability of the free film interface.

3 Rotating films are typically very thin, and as such, wall roughness is believed to destroy the laminar portion of the thermal boundary layer. Thus, common incipient boiling models based on the existence of a linear temperature profile in the vicinity of the heated surface should be avoided if the surface fails to satisfy the smoothness condition of equation (9). The present data also indicate the existence of a different mechanism for boiling incipience that may be the result of turbulent exchange of heat between the wall and the bulk of the film rather than molecular diffusion.

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Limits to Critical Heat Flux Enhancement in a Liquid Film Falling Over a Structured Surface That Simulates a Microelectronic Chip

T. A. Grimley,[1,2] I. Mudawwar,[1] and F. P. Incropera[1]

Nomenclature

L = length of heater
q = heat flux based on the total base area of the enhanced surface
q_M = critical heat flux (CHF) based on the total base area of the enhanced surface
T = temperature
ΔT_{sat} = wall superheat at the base of the enhanced surface = $T_w - T_{sat}$
ΔT_{sub} = inlet subcooling = $T_{sat} - T_{in}$
U = mean inlet velocity
δ = film inlet thickness

Subscripts

in = inlet
sat = saturated
sub = subcooled
w = wall

[1]School of Mechanical Engineering, Purdue University, West Lafayette, IN 47907.

[2]Fluid and Thermal Systems Group, Southwest Research Institute, San Antonio, TX 78284.

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Introduction

The combination of increasing power densities and stringent surface temperature constraints for microelectronic components has stimulated interest in cooling by means of pool boiling in a dielectric liquid. However, although dielectric fluorocarbons such as FC-72 (manufactured by 3M) are highly compatible with electronic components, small thermal conductivities and latent heats make them poor heat transfer fluids. Moreover, pool boiling studies concerned with CHF enhancement (Bergles and Chyu, 1982; Marto and Lepere, 1982) have shown that significant hysteresis (temperature overshoot) at the inception of boiling may violate component temperature limitations.

More recently, the falling liquid film has been considered as a means of enhancing the performance of dielectric coolants by controlling boiling hysteresis and increasing the critical heat flux (CHF). From experimental studies performed for a gravity-driven liquid (FC-72) film flowing over a smooth surface (Mudawwar et al., 1987), the ability to suppress hysteresis was demonstrated, and the trend of increasing CHF with decreasing heater length was determined. However, even for the smallest heater length of the study (L = 12.7 mm), CHF values were not much higher than those associated with pool boiling.

To determine whether structured surfaces could be used to substantially extend CHF, Grimley et al. (1987) performed experiments for a thin film of FC-72 falling over a 63.5-mm-long surface with either longitudinal microfins or microstuds. Although both the microfin and microstud surfaces enhanced nucleate boiling heat transfer relative to a smooth surface, only the microfin surface provided significant enhancement of CHF. CHF was observed to be due to dryout of a thin subfilm, which remained on the boiling surface after the bulk of the fluid in the falling film had separated due to intense vapor generation. It was argued that the microfins extended CHF by allowing surface tension forces to maintain the liqud film on the surface more effectively and by inhibiting the lateral spread of dry patches after film separation. In contrast, the microstud surface acted to break up the film, thereby hastening film separation and decreasing CHF. In addition, it was found that CHF could be enhanced by subcooling the liquid or by installing a louvered flow deflector a short distance from the heated surface. While subcooling decreased the intensity of vapor effusion by supplying the heated surface with liquid of reduced temperature, the deflector inhibited film separation.

On the basis of the foregoing results, it is known that CHF in a falling liquid film may be enhanced by reducing the length of the heated surface, machining longitudinal grooves in the surface, shrouding the surface with a louvered flow deflector, or subcooling the liquid. However, experiments have yet to be performed in which these effects are considered collectively in